

Analysis of the Effect of Construction and Operation of Thermal Expansion System Compounds on Steam Turbines Reliability

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Abstract: The inspection results are presented of turbines of different types and capacity, showing the influence of various factors (such as increased frictional forces on the mating surfaces, clearance changes in the joints elements, TES elements design, state of the thermal expansions compensation system of pipelines) on the operation both of thermal expansion system and of the turbine as a whole. The data are presented on the effectiveness of various measures aimed to eliminate the causes of the turbine thermal expansion system deviations from its normal operation.

The results are shown of the influence simulation of various factors (such as flanges and piping warming, ratio of clearance changes in the elements) on the probability of turbine TES hindrance. It is shown that clearance ratios employed in most turbines do not provide the stability of turbine TES against the external action of connected pipes. The simulation results permit to explain the bearing housings turns observed during inspections, resulting in a jam on the longitudinal keys, in temperature distribution changes on the thrust bearing pads, and in some cases in false readings of instruments rotor axial displacement.

Reliability study results of steam turbine plants equipment operation show [1] that the thermal expansion system (TES) of the turbine is one of the critical (most damageable) components of multi-cylinder turbines. When turbine operation age after the repairs reaches 25—30 thousand hours the malfunctions in TES operation lead to defects both in the flowing part and in the bearings. The situation is complicated by a significant period of operation (on the verge of resource exhaustion) of most of the turbines installed at the power stations of the Russian Federation, by turbine operation variable schedule along with a decrease in the amount of repair work. In this regard, for most of the turbine units in operation a priority is to solve the problem of ensuring the operation of the TES, both in terms of the impact on reliability and on the economy of their future operation.

To improve the reliability of the TES some leading industry institutes and organizations have developed and implemented a number of measures [2] aimed at eliminating the main causes of thermal expansion hindrance in the bearing housings. It should be noted that the new turbine units on which the measures recommended by [2] are implemented also have problems related to the functioning of TES. So, for example, only in 2016, the authors carried out works to determine the causes of the hindered thermal expansion of two new turbines, mounted in 2013 and 2016. The revealed causes of malfunction accounted, in the authors' opinion, for insufficiently clear and accurate knowledge about the processes taking place in TES.



To investigate these processes, the authors proposed a generalized chart shown in Fig. 1 and described in detail in [3]. The chart presents an interaction of an optional cylinder of a multi-cylinder steam turbine, of the bearing housing connected with it (on the regulator side), and of the guiding longitudinal keys as the foundation elements.

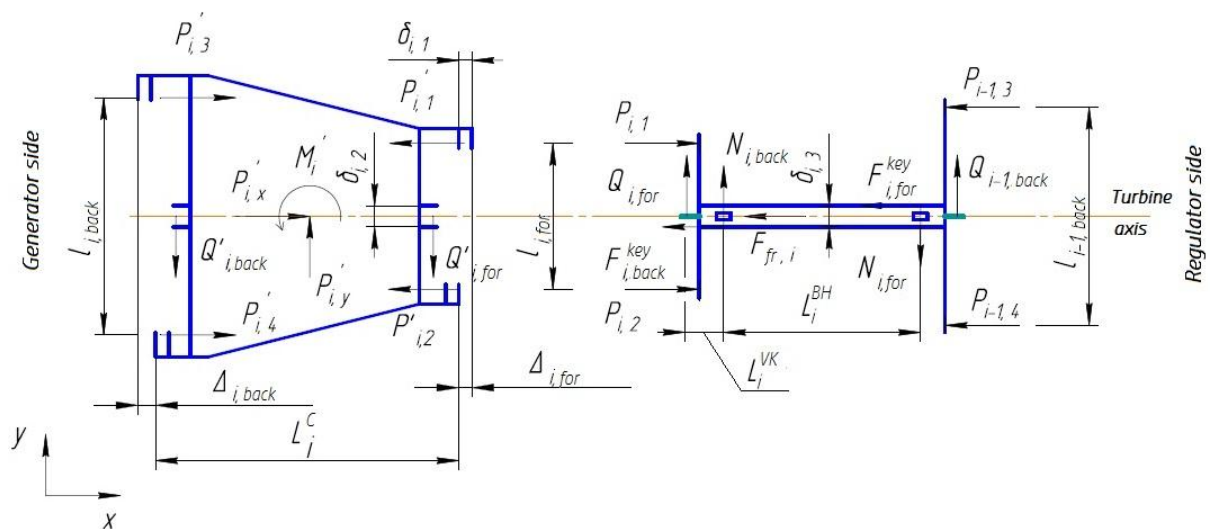


Fig. 1 Generalized chart of the interaction of an optional cylinder of a multi-cylinder turbine with the bearing housing connected with it

On the basis of the presented generalized scheme, two models were developed:

- **an analytical model** for the interaction of the thermal expansion system elements in case of a temperature nonuniformity, and in particular for determining the values of the frictional forces arising on longitudinal keys;
- **kinematic model** of the turbine cylinder joint with bearing housing and of the bearing housing with foundation frame.

In the analytical model the following simplifications and assumptions were made:

1. external forces, including those acting on the turbine from the pipelines connected to it, are not taken into account;
2. the weight load resultant of the cylinder and the bearing housing on the cross-bar lies in the vertical plane XZ passing through the axis of the turbine;
3. the axial stiffness of the protrusions ("wings") of the bearing housing is the same for all bearing housings;
4. the transverse keys clearances are absent (equal to zero);
5. stresses that occur on vertical keys are not taken into account;
6. clearance changes on the longitudinal key occurs symmetrically when the bearing housing is rotated.

The system of equations for simulation of axial forces values on the turbine cylinder feet and of the forces on the longitudinal keys in case of temperature nonuniformity is compiled on the basis of the design scheme, proceeding from the equilibrium conditions of the cylinder, from the equilibrium of the bearing housing, and from the compatibility of deformations and accepted assumptions.

Based on the analytical model developed, an analysis was performed for a two-cylinder turbine with dimensions of cylinders and bearing housings close to the dimensions of the high and low pressure cylinders of T 100 / 120-130 family turbines produced by the Turbomotor Works (ZAO Ural Turbine

Works at present) and with corresponding weight loads on the front and middle bearings housings. The analysis showed the following:

- with a 0.05 mm clearance on the longitudinal keys [4], the contact between the front bearing housing (BH1) and longitudinal keys arises at a temperature nonuniformity of slightly more than 2 °C; for the middle bearing housing (BH2) a contact with longitudinal keys occurs at a temperature nonuniformity of about 5 °C;
- when the temperature difference permissible according to the factory instruction manual [5] exceeds 10 °C, the force acting from the front bearing housing on each longitudinal key increases by more than 1.5 times — from 147 kN (15,000 kgf) to 245 kN (25,000 Kgf);
- with a temperature difference of 20 °C, which is often observed during turbine start-up operations, the total frictional force increases more than twice, reaching a value of 367 kN (37 440 kgf).

It should be noted that the real picture, by virtue of the assumptions made in the model, may differ slightly from the one calculated by the proposed model: the presence of clearances on the transverse keys increases the amount of temperature nonuniformity at which a contact arises between the bearing housing and the longitudinal key. On the contrary, the angular movements of cylinder feet against the bearing housings, caused by the peculiarities in thermal state of the cylinders [6], lead to an earlier occurrence of contact. Some effect on the total picture will also be provided by contacts on vertical keys.

To verify the analysis, a model was also developed for performing finite element analysis (FEM).

Based on the results of calculations by both the analytical and the FEM models the diagrams of the effect of temperature nonuniformity on the flange of the high pressure cylinder horizontal split were plotted against the frictional forces total values variations on the joint surfaces of the front bearing housing with the base frame. It was revealed that the analytical model gives excessive values of forces applied to the longitudinal keys within the range of temperature nonuniformity from 2.5 to 20 °C (for the temperature nonuniformity of 10 °C the deviation is 40 %, for 20 °C — 16 %). When the temperature nonuniformity on the flange of the high pressure cylinder horizontal split exceeds 20 °C, the difference between the values of the total frictional force decreases (at 30 °C the difference amounts to 6 %, and at 45 °C the values practically coincide).

The relationship between the value of the longitudinal key clearance and the permissible temperature nonuniformity, in which there is still no contact in the pair "longitudinal key-groove", is linear. Thus, to avoid contact at the permissible temperature nonuniformity of 10 °C, the gap on the longitudinal keys should not be less than 0.25 mm. As to the turbine operation reliability, the rise of this gap to 0.25 mm should not lead to any complications. With a distance between the extreme points of the longitudinal keys under the front bearing housing being of 1500 mm, such a gap will ensure the rotation of the housing in the horizontal plane, equivalent to a change in the vertical plane slope of 0.17 mm/m, which is less than the permissible value of turbine swing in the vertical plane about 0.2 mm/m for turbine shutdown.

It should be noted that in analyzing the above reasons for jamming in the pair "longitudinal key — groove" the turbine cylinder was considered as stationary or moving strictly along the axis of the turbine unit. The studies presented in [7, 8] showed that the cylinders of the turbine are subjected to forces and moments from the attached pipelines, the values of which can be sufficiently large. These forces and moments can shift the cylinders of the turbine relative to the bearing housings, and shift the bearing housings relative to the turbine axis within the limits imposed by gaps in the keyed joints of TES. This is confirmed by the results of field studies carried out by the authors both on turbines for a long time in operation and on turbines of different manufacturers recently commissioned.

Thus, on a single-cylinder backpressure turbine with a capacity of 20 MW, which has been in operation for a long time, the rotation of the bearing housing in the horizontal plane reached values that lead not only to the jamming of the bearing housing on the longitudinal keys, but also to incorrect readings of the axial shear sensor located in the plane of the horizontal split. Considering the small length of the turbine cylinder and the relatively low temperature of the hot steam (400 °C), and the

gaps in the transverse and longitudinal keyed joints are much higher than the normative ones, the most likely cause of the bearing housing swing is the rotation of the cylinder under the action of the attached pipelines.

Investigations of the recently commissioned multi-cylinder turbine K 330-24.5 at a power station with supercritical steam parameters, where there had been implemented project measures to exclude jamming in transverse keyed joints (rotary keys had been installed along with antifriction modules from the composite material applied on the sliding surface of bearing housing), also showed that the cylinders and bearing housings rotated relative to the axis of the turbine (Fig. 2).

Analysis of the above mentioned kinematic model of interaction between the cylinder of the turbine, the bearing housing and the foundation frame under the action of shear forces and moments from the attached pipelines, developed on the basis of a generalized scheme, showed that when the cylinder or bearing housing are affected by the attached pipelines 4 types of contact are possible in the pair "longitudinal key — groove" (Fig.3):

- a) one-sided contact with both longitudinal keys;
- b) contact with only the most distant from the cylinder longitudinal key;
- c) contact with only the closest to the cylinder longitudinal key;
- d) "diagonal" contact with both longitudinal keys.

The first three types of interaction between the groove of the bearing housing and the foundation frame can be combined into a group of "one-sided" contacts, under which additional frictional forces can occur at the points of contact between the groove and the longitudinal keys, but the bearing housing jamming on the longitudinal key is excluded.

The most unfavorable for the TES functioning is the "diagonal" type of contact (Fig. 3, d). In this case the probability of jamming the bearing housing on longitudinal keys is the highest. It is for this

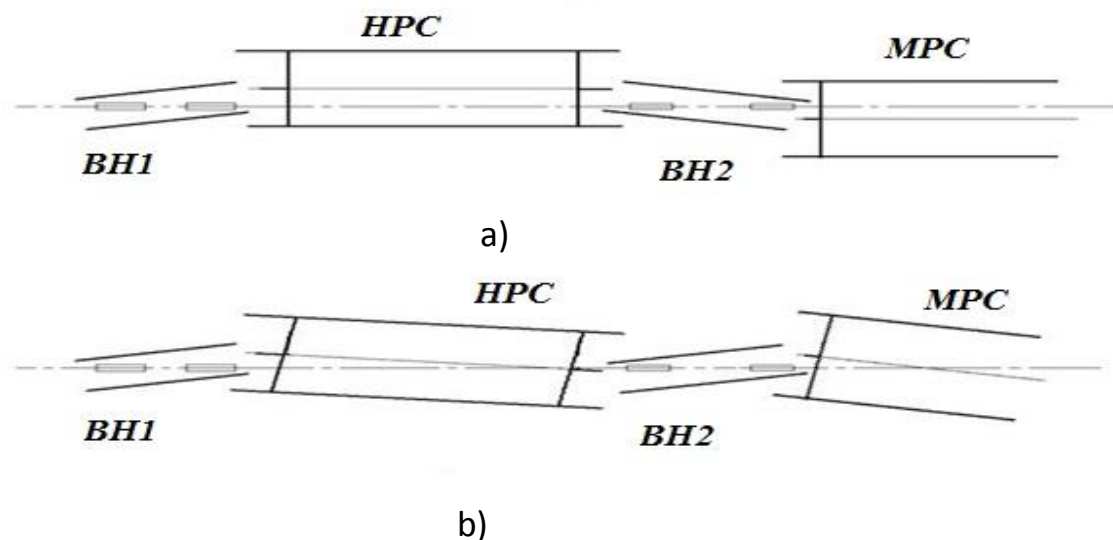


Fig. 2 Variations in the mutual position of the cylinders, bearing housings and guides on the base frames of the 330 MW turbine:

a) at start; b) after warming-up

type of contact that an interaction analysis was made by finite element method, based on the developed model.

Rotation of the bearing housing with respect to the longitudinal keys can occur either as a result of the turbine cylinder rotation in the horizontal plane relative to a certain center, or in the case of a plane-parallel displacement of the cylinder across the axis of the turbine unit.

When formulating the kinematic model and performing the analysis, the following assumptions were made:

- original gaps in the "keyed joints" are set up symmetrically;
- turbine cylinder and the bearing housing can rotate only relative to the centers O_c and OKP , which are the centers of the segments L_1^C and L_1^{BH} , respectively;
- rotation of the turbine cylinder its transverse displacement relative to the axis of the turbine unit are taken into account separately;
- mutual angular deformations of the turbine cylinder feet and supporting "wings" of the bearing housing were not taken into account.

The boundary conditions for the rotation of the turbine cylinder, in which there can not arise a "diagonal" contact on the longitudinal keys, can be written as follows

$$\frac{\delta_{1,1}}{\delta_{1,3}} < \frac{\delta_{1,2}}{\delta_{1,3}} \times K_1 + (1 + 2K_2) \times K_1, \quad (1)$$

where K_1 and K_2 are dimensionless coefficients that depend on the geometric characteristics of the junction of the turbine cylinder with the bearing housing and the foundation frame, determined from expressions:

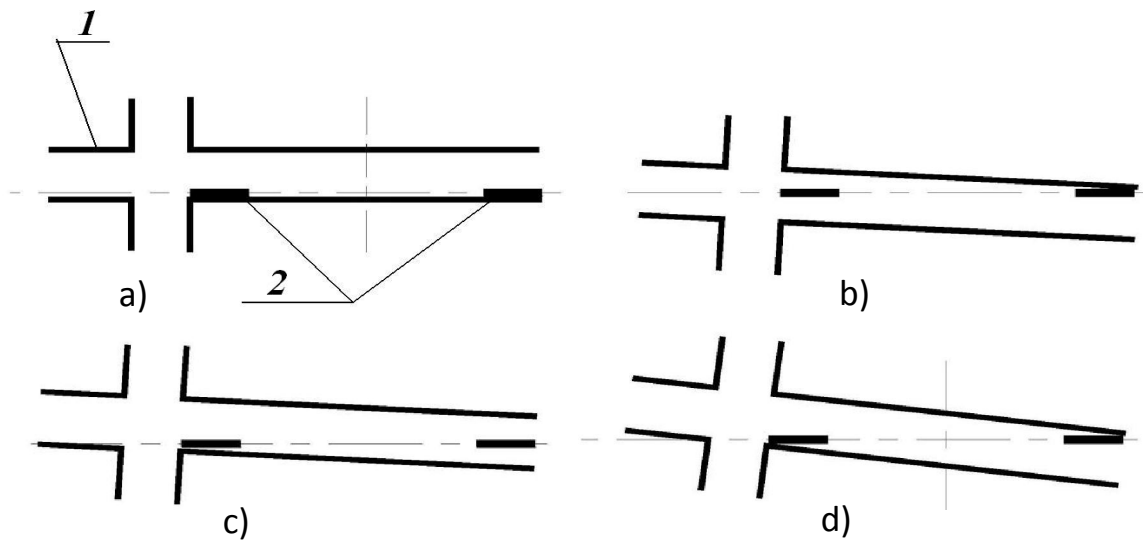


Fig. 3. Interaction schemes between the bearing housing and the foundation frame:
1 – bearing housing. 2 – longitudinal keys

$$K_1 = \frac{l_{1,for}}{L_1^C};$$

$$K_2 = \frac{L_1^{VK}}{L_1^{BH}}.$$

With a plane-parallel displacement of the turbine cylinder across the axis of the turbine unit, a "diagonal" contact between the longitudinal keys and the bearing housing cannot occur under the following boundary conditions:

$$\frac{\delta_{1,1}}{\delta_{1,3}} < K_3. \quad (2)$$

where $K_3 = \frac{L_1^{BH}}{l_{1,for}}$.

Employing the boundary conditions (1) and (2), the boundary of TES stability against the appearance of a "diagonal" contact in the pair "longitudinal key — groove" or the system stability boundary to the external action is described by the expression

$$\frac{\delta_{1,1}}{\delta_{1,3}} = \min \left(\left(\frac{\delta_{1,2}}{\delta_{1,3}} \times K_1 + (1 + 2K_2) \times K_1 \right), K_3 \right) \quad (3)$$

One can estimate the stability of the turbine cylinder TES to the external effect, using the "stability diagram" provided by the authors (Fig. 4). On the abscissa of the diagram the ratio is set of the gaps on the vertical key to the gaps on the longitudinal keys $\frac{\delta_{1,2}}{\delta_{1,3}}$, against the ratio of the gaps on the transverse key to the gaps on the longitudinal key $\frac{\delta_{1,1}}{\delta_{1,3}}$. On the diagram field a stability boundary is plotted from the expression (3). The stability boundary divides the field of the diagram into two areas:

- the area of the system stability is below the stability boundary, when inequalities (1) and (2) are satisfied;
- the area of the unstable state of the system is above the stability boundary when at least one of the inequalities (1) and (2) is not satisfied.

Thus, if the point corresponding to the current state of the gaps in the turbine cylinder TES lies above the stability boundary, then the external influence can negatively affect TES functioning, i.e. can cause thermal expansion hindrance of the bearing housing.

When designing new turbines the intervals of permissible values are usually specified for clearances in TES keyed joints, which can be described by the system of inequalities:

$$\begin{cases} \delta_{1,1}^{min} \leq \delta_{1,1} \leq \delta_{1,1}^{max} \\ \delta_{1,2}^{min} \leq \delta_{1,2} \leq \delta_{1,2}^{max} \\ \delta_{1,3}^{min} \leq \delta_{1,3} \leq \delta_{1,3}^{max} \end{cases},$$

where $\delta_{1,1}^{min}$, $\delta_{1,2}^{min}$, $\delta_{1,3}^{min}$ stand for minimal gap values on vertical, transverse and longitudinal keys respectively;

$\delta_{1,1}^{max}$, $\delta_{1,2}^{max}$, $\delta_{1,3}^{max}$ stand for maximal gap values on vertical, transverse and longitudinal keys respectively.

In the "stability diagram" coordinate system all possible combinations of the TES gaps are located inside the "region of normative clearances", which is specified by the system of inequalities

$$\begin{cases} \frac{\delta_{1,2}}{\delta_{1,3}} \times \frac{\delta_{1,1}^{min}}{\delta_{1,2}^{max}} \leq \frac{\delta_{1,1}}{\delta_{1,3}} \leq \frac{\delta_{1,2}}{\delta_{1,3}} \times \frac{\delta_{1,1}^{max}}{\delta_{1,2}^{min}} \\ \frac{\delta_{1,2}^{min}}{\delta_{1,3}^{max}} \leq \frac{\delta_{1,2}}{\delta_{1,3}} \leq \frac{\delta_{1,2}^{max}}{\delta_{1,3}^{min}} \\ \frac{\delta_{1,1}^{min}}{\delta_{1,3}^{max}} \leq \frac{\delta_{1,1}}{\delta_{1,3}} \leq \frac{\delta_{1,1}^{max}}{\delta_{1,3}^{min}} \end{cases}$$

To estimate TES stability to the external effect for the cylinder in the design one should build an "area of normative gaps" in the field of the "stability diagram".

Fig. 4 presents an example of TES stability diagram for high pressure cylinders of the turbines by TMZ manufacturing. It can be seen that normative clearance areas gaps are either entirely outside the area of the system stability (i.e., above the stability boundary), or only a part of them belongs to the stability area.

An analysis of TES gaps obtained by the authors in surveying the turbines with problems of thermal expansions during start-ups and shutdowns, carried out according to the above presented method, showed that in all cases the point corresponding to the current state of the TES gaps was in the area of the unstable state of the system.

In order to ensure the TES stability of the high pressure cylinders, for example for turbines of the T-100 / T-110 family, the ratio of the gap in the transverse keys to the value of the gap on the longitudinal keys should not exceed 0.8. That means that with a normative clearance on the longitudinal keys of 0.06 mm, the clearance on the transverse keys should not exceed 0.048 mm (with

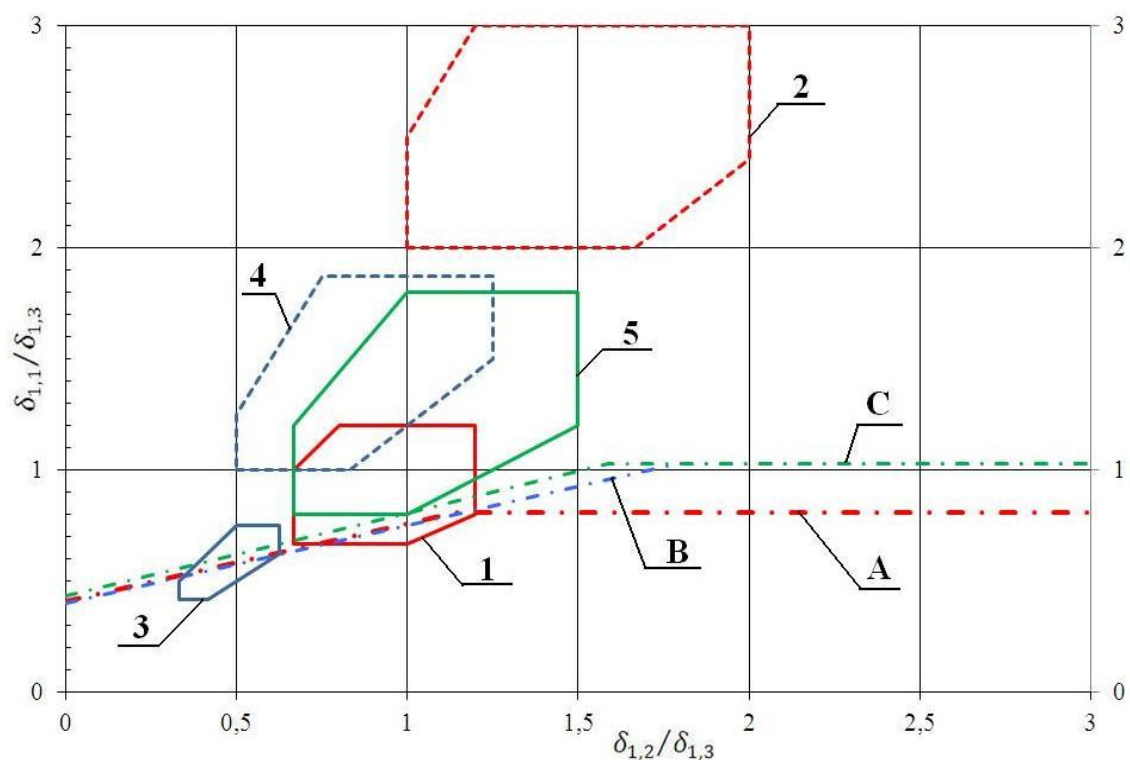


Fig. 4. Stability boundaries and areas of normative clearances for some turbines manufactured by TMZ

A – stability boundary of T-100/T-110 high pressure cylinder; B – stability boundary of T-175/T-185 high pressure cylinder; C – stability boundary of T-250 high pressure cylinder.

1 – area of normative clearances of T-100 high pressure cylinder; 2 – area of normative clearances of T-110 high pressure cylinder; 3 – area of normative clearances of T-175 high pressure cylinder; 4 – area of normative clearances of T-185 high pressure cylinder; 5 – area of normative clearances of T-250 high pressure cylinder

a minimum normative clearance of 0.04 mm). The operational experience of [6, 9] shows that gradually in connection with the cylinders temperature deformations, an uncontrolled increase of gaps occurs in the standard transverse keyed joints. At the same time, if the value of the clearance on the longitudinal keys, as mentioned earlier in connection with minimizing the consequences of temperature distortions on the flanges of the turbine cylinder, is increased to 0.20...0.25 mm, then the value of the gap on the transverse keys should not exceed 0.20 mm.

The minimum value of the gap on the vertical key is determined from the expression (1) and for the assumed gaps ratio on the transverse and longitudinal keys the value of the gap ratio on the vertical

key to the gap on the longitudinal key should not exceed 1.03. Accordingly, the size of the gap on the vertical key should not be less than 0.26 mm, which is significantly higher than the currently accepted gaps on vertical keys. If one assumes the value of the maximum gap on the vertical key of 0.10 mm along with the clearance on the longitudinal key of 0.20 mm, then the gap on the transverse keys should not exceed 0.12 mm.

Since the value of the gap on the transverse keys for the example in question is the value that determines the immunity of the system to external effect, and this single joint which can be monitored and adjusted during the interrepair period, the task of monitoring the state of this joint and preserving its parameters becomes very important in terms of ensuring the normal thermal expansion of steam turbines cylinders.

Conclusions:

1. The stability of steam turbine system of thermal expansion to an external effect depends to a large extent on the ratio of the gaps in the longitudinal, transverse and vertical keyed joints.
2. To increase the stability of the thermal expansion system against temperature distortions, it is advisable to increase the clearance on the longitudinal keys to a value that could provide a rotation in the horizontal plane of about 0.2 mm/m.
3. To ensure the stability of thermal expansion systems for steam turbines in operation, in the course of overhaul it is advisable to set the gaps in the transverse keyed joints according to the results of calculations based on real gaps data of longitudinal and vertical keyed joints.
4. The design of transverse key joints should ensure the possibility of monitoring their state (gap values), high maintainability (the possibility of restoring clearances) during the interrepair period or to ensure the constancy of the specified gaps (for example, separation plates or rotary and split transverse keys).

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